



Energy, Exergy and Economic Analyses of Different Configurations for a Combined HGAX/ORC Cooling System

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Abstract

A comparative study is performed to investigate the improved heat-driven refrigeration systems. The systems use a low-temperature heat source to produce low-temperature cooling ($-55\text{ }^{\circ}\text{C}$). The base system consisted of a hybrid GAX (HGAX) cycle and a Rankine cycle. Three major features for the HGAX cycle are proposed in four configurations to reduce the high temperature of the compressed fluid exiting the compressor. The operating parameters of the systems having the maximum exergy efficiency are computed, and the corresponding performance parameters are applied in all configurations. The results show that the best configuration that has higher exergy efficiency is the one utilizing two compressors in the HGAX cycle. For the heat source temperature of $133.5\text{ }^{\circ}\text{C}$, this configuration has 34.7% higher energy utilization factor, 33% higher exergy efficiency, 11% lower total product cost, and 28% lower circulating cooling water of the cooling tower than the base system.

Keywords Refrigeration · Improved HGAX cycle · Low-temperature heat source · Low-temperature cooling · Rankine cycle

Abbreviations

AHE	Auxiliary heat exchanger
AGAX	Absorber GAX
\dot{C}	Cost rate (\$/s)
COP	Coefficient of performance
CRF	Capital recovery factor
DGAX	Desorber GAX
e	Efficiency
\dot{E}	Exergy rate (kW)
EI	Energy index
EUF	Energy utilization factor
GAX	Generator–absorber heat exchange (already defined)
h	Enthalpy
HGAX	Hybrid GAX
i	Interest rate
\dot{m}	Mass flow rate (kg/s)
n	System life (year)
P	Pressure (kPa)
\dot{Q}	Heat transfer rate (kW)

RC	Rankine cycle
RHX	Reheated heat exchanger
T	Temperature ($^{\circ}\text{C}$)
\dot{W}	Power rate (kW)
Z	Investment cost (\$/s)

Greek symbols

φ	Maintenance factor
τ	Annual operation hours

Subscripts

a	Absorber
ava	Available
B	Boiler
Com	Compressor
CT	Cooling tower
CW	Circulating water in the cooling tower
eva	Evaporator
ex	Exergy
G	Generator
geo	Geothermal
HS	Heat source
hw	Hot water
in	Input
out	Output
P	Product
tur	Turbine
req	Requirement

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1 Introduction

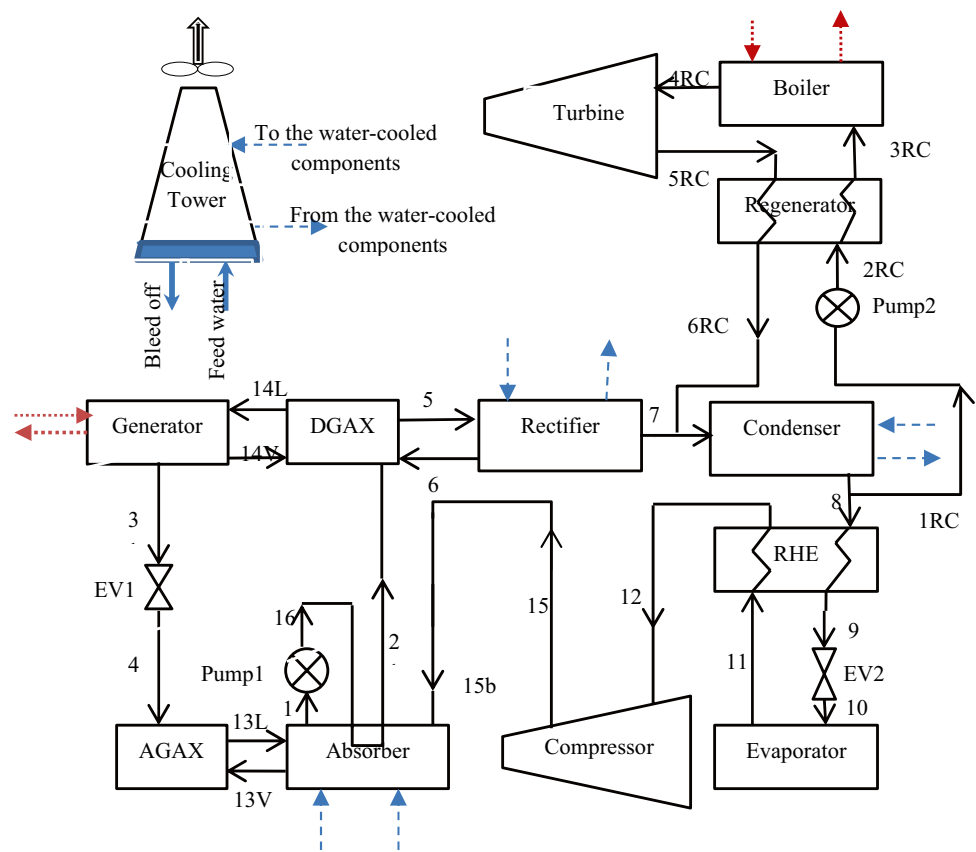
It is a major worldwide concern to reduce the fuel consumption and save energy by using more efficient systems. For example, in the low-temperature refrigeration applications, improving the energy efficiency of systems is noteworthy, while in these applications the energy efficiency of traditional absorption refrigeration systems is low and the power consumption of compression refrigeration systems is high. To have better performance at low-temperature applications, two-stage absorption systems and two-stage compression systems were used and improved. Rogdakis and Antonopoulos (1992) studied a two-stage ammonia–water absorption refrigeration system that provides low-temperature cooling. This system operated at three pressure levels, and its coefficient of performance (COP) varied from 0.20 to 0.65 at evaporating temperatures of -70 to -30 °C for 10 °C ambient temperature. The disadvantages of such systems are the low COP and large heat dissipation (Du et al. 2017). Du et al. (2017) improved a two-stage ammonia–water absorption refrigeration system for low evaporating temperatures. They maximized the interval heat recovery by pinch technology. Their results show that the COP of the new configuration is 14.5% and 34.1% higher than the conventional system at the evaporation temperatures of -10 °C and -30 °C, respectively. When there is a temperature overlap between the generation and absorption, the improved system is more efficient. This system, however, has a higher initial cost, which is a disadvantage. A CO₂/NH₃ cascade refrigeration system consisting of two compression cycles is a common system to produce low-temperature cooling. This system is analyzed and used by many researchers (Lee et al. 2006, Rezayan and Behbahaninia 2011). However, this system used a large amount of power that was considered as a disadvantage. Baek et al. (2005) tried to reduce the power consumption of two-stage compression systems. They analyzed a transcritical carbon dioxide cycle with two-stage compression and intercooler cycle. They showed that the COP of the intercooler cycle is up to 25% larger than the COP of the basic single-stage transcritical cycle. Hybrid and cascade refrigeration systems are used widely to produce low-temperature cooling. Anand et al. (2014) compared thermodynamically a hybrid refrigeration system suitable to operate as absorption system, compression–absorption system and compression system. They showed that compressor work and exergy loss are lesser for compression–absorption mode when compared to compression mode (Anand et al. 2014). A cascade absorption/compression refrigeration system was analyzed by Gari-mella et al. (2011). This system provided cooling at temperatures as low as -40 °C. The power consumption in

this system was 31% lower than a compression refrigeration system under the same operating conditions. But the integrated systems that used both absorption and compression cycles for refrigeration applications make the system complex.

An absorption cycle with a generator–absorber heat exchanger (GAX) can improve the COP of the absorption refrigeration system by 20–30% (Mehr et al. 2013). Compared with the double effect cycles, the GAX cycle decreases pumping input and is more flexible than the multiple-stage cycles (Xu and Wang 2016). The GAX cycle can be improved to more efficiently products. A novel GAX absorption refrigeration cycle was proposed by Shi et al. (2016). Their system used the absorption heat that was not used by simple GAX cycle, and its COP was 20% higher than the simple GAX cycle. At low evaporator temperatures, the overlapping temperature range between the generator and the absorber is vanished and the GAX cannot be used. The hybrid GAX (HGAX) cycle by utilizing a compressor between evaporator and absorber can provide higher absorber pressure and provided low-temperature cooling. Kang et al. (2004) improved four different types of HGAX cycles for different applications. Their results showed that low-temperature application type can produce temperatures as low as -80 °C with COP of 0.3. In addition, the COP of the HGAX cycle is about 30% higher than that of simple GAX cycle at the same operating conditions (Kumar and Udayakumar 2007). Therefore, in this research the hybrid GAX cycle is preferred to study and analyzed for low-temperature applications.

To eliminate the power consumption of the refrigeration systems and utilizing totally heat-driven systems, the power cycles can be integrated with the refrigeration cycles and produce the power requirements of the refrigeration systems. Power cycles with organic or multi-component working fluids could produce power from low-temperature heat sources. Thermodynamic analysis of two models of a tri-generation system was carried by Zare (2016). These systems utilized low-grade geothermal energy at 120 °C to producing cooling, heating and power. An absorption refrigeration cycle and a water heater were combined by an organic Rankine cycle (ORC) in one model and by a Kalina cycle in the other system. Mohammadi et al. (2019) proposed a multi-generation system for generating power and triple effect refrigeration at different evaporation temperatures. They utilized an ORC and a GAX cycle to recover the waste heat of a Bryton cycle that combines with a dual-evaporator cascade carbon dioxide–ammonia compression refrigeration system. They showed that their system provided cleaner production of all products with improved efficiency, economics. But the deficiency of the exergy analysis is noticed in their study. Parikhani et al. (2020) investigated a new ammonia–water mixture CCHP system that used a low-temperature heat

Fig. 1 Schematic diagrams of the base system



source and a modified version of a Kalina cycle. The temperature of the cooling product of their system is above $0\text{ }^{\circ}\text{C}$. Mousavi et al. (2019) studied on their proposed cascade absorption–compression refrigeration system from the view point of exergy, exergoeconomic, and exergoenvironmental. In their system, a CO_2 compression cycle is utilized as the subsystem to produce refrigeration at $-54.62\text{ }^{\circ}\text{C}$. An absorption cycle is used to produce cooling of the condenser of the subsystem. The temperature of the heat input to the system is $350\text{ }^{\circ}\text{C}$.

The authors (Seyfour et al. 2018) used a low-temperature heat source to produce cooling at low temperatures for the first time. They proposed a system combined of a Rankine cycle (RC) and a HGAX cycle. Their system produced cooling at $-50\text{ }^{\circ}\text{C}$, using a low-temperature heat source at $133.3\text{ }^{\circ}\text{C}$. But when the HGAX cycle is utilized for low-temperature applications and the pressure ratio of the compressor is high, the temperature of the working fluid after the compression process is high. To reduce this temperature for lowering water consumption in cooling tower and also to upgrade the system performance, four improved configurations of the Seyfour et al.'s (2018) system are presented in this paper. These configurations offer three major novelty features that can increase the energy and exergy efficiencies of the system. The Seyfour et al.'s (2018) system is preferred as the base system. The heat

source of the system is low-temperature geothermal water. The COPs, energy and exergy efficiencies, total product costs and circulation water of the cooling tower of these configurations are compared with the base system. The novelty of this study is the plan of a new system that offers more efficiently and economically production of cooling. This system should prepare some advantage to the related literature and industrial studies.

2 Modeling and Assumptions

The main parts of the base system are Rankine power cycle and HGAX cycle with a cooling tower supplying the coolant water for water-cooled components as shown in Fig. 1. The Rankine power cycle utilizes the hot water from the heat source and produces enough power to run the refrigeration cycle. The power and refrigeration cycles used a single condenser, and both the mass and energy are exchanged between them. The ammonia–water mixture is used as the working fluid. The complete system is modeled using the software package Engineering Equation Solver (EES) (2013). The mass, energy and exergy balance equations are solved for all components of the system. In order to simulate the system in this work, the following assumptions are made:

- The ambient pressure and temperature are assumed to be 101 kPa and 25 °C, respectively.
- The exergy of a stream is the sum of the physical and chemical exergy of the mixture.
- The total circulating water in the cooling tower is the sum of the cooling water of the condenser, intercooler, absorber and rectifier.
- The relative humidity of the ambient air is 30%.
- The difference between the cooled water temperature in the cooling tower and the wet bulb temperature is 5 °C (Mohammadi and Ameri 2014).
- The liquid-to-gas mass ratio (LG) for cooling tower is 1.2 (Mohammadi and Ameri 2014).
- To calculate the fan power consumption of the cooling tower, $\dot{W}_{fan,CT}$ the following equation is used (Saidi et al. 2011):

$$EI = \frac{\dot{Q}_{CT}}{\dot{W}_{fan,CT}} \quad (1)$$

EI is the energy index and \dot{Q}_{CT} denotes the cooling tower capacity.

- The energy index is 0.2 (Saidi et al. 2011).

As mentioned before, the power requirement of the HGAX cycle is produced by the Rankine cycle, no additional power is added to the system, the only input of the system is the geothermal hot water and the only output of the system is the cooling load. The coefficient of performance of the refrigeration cycle (HGAX cycle) is defined as the cooling load to the HGAX cycle input energy. The input energy of the HGAX cycle is the heat requirement of the generator plus the power requirement of the pumps and compressor.

$$COP = \frac{\dot{Q}_{eva}}{\dot{Q}_G + \dot{W}_{pump} + \dot{W}_{com}} \quad (2)$$

The energy utilization factor (EUF) as the energy efficiency of the overall system is the cooling load, as the only product of the system, divided into the input energy of the system. The input energy of the total system is the heat input the generator plus the heat requirement of the boiler:

$$EUF = \frac{\dot{Q}_{eva}}{\dot{Q}_G + \dot{Q}_B} \quad (3)$$

\dot{Q}_B and \dot{Q}_G are the input energy of the boiler and generator, respectively. The exergy efficiency of the overall system is defined as the ratio of the exergy of the cooling load to the exergy input into the generator and boiler:

$$e_{ex} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \frac{\dot{Q}_{eva} \left(1 - \frac{T_0}{T_{eva} + \Delta T_{pinch}} \right)}{\dot{E}_{in}} \quad (4)$$

ΔT_{pinch} is the pinch temperature in the evaporator. \dot{E}_{in} is the exergy of the input hot water to the system from the geothermal heat source.

$$\dot{E}_{in} = \dot{m}_{hw} [(h_{hw,in} - h_{hw,out}) - T_o (S_{hw,in} - S_{hw,out})] \quad (5)$$

2.1 Cost Balance

Total expenditures to generate the products of a system can be obtained from the cost balance equation for the overall system (Akbari and mahmudi 2017; Sayyadi and Nejatollahi 2011):

$$\dot{C}_{p,total} = \dot{C}_{fuel} + \sum_{k=1}^{n_k} \dot{Z}_k \quad (6)$$

$\dot{C}_{p,total}$ is the total product cost, $\sum_{k=1}^{n_k} \dot{Z}_k$ is the total capital cost of the system and \dot{C}_{fuel} is the cost of the fuel that for the geothermal resource (Seyfour et al. 2018):

$$\dot{C}_{fuel} = c_{geo} \times \dot{E}_{in} \quad (7)$$

c_{geo} denotes the unit cost of geothermal water that is the exploration and drilling cost of the geothermal resource and calculated as follows (Seyfour et al. 2018):

$$c_{geo} = \frac{2000 \exp(-0.0045(\dot{Q}_{geo} - 5)) \times CRF}{\dot{m}_{geo} \times h_{geo} \times \tau} \quad (8)$$

\dot{Q}_{geo} and h_{geo} are the geothermal water heat capacity in MW and the energy value of geothermal water, respectively. \dot{Z}_k in the relation (5) is the investment and maintenance cost of a component of the system (Seyfour et al. 2018):

$$\dot{Z}_k = Z_k \cdot CRF \cdot \frac{\varphi}{\tau \times 3600} \quad (9)$$

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (10)$$

φ denotes the maintenance factor ($\varphi = 1.1$), CRF is the capital recovery factor and τ indicates the annual operation hours of the system; also, i and n denote the interest rate ($i = 15\%$) and system life ($n = 20$ year), respectively (Seyfour et al. 2018). To calculate the capital cost for a component of the system (Z_k), a relation associated with that component is used as the following equations (Seyfour et al. 2018; Ifaei et al. 2016):

Table 1 Reference costs of heat exchangers (Seyfour et al. 2018)

Component	Reference cost (\$)
Generator	17,500
Absorber	16,500
Rectifier	12,000
Condenser	8,000
Boiler	17,000
RHX	12,000
Evaporator	16,000
Regenerator	12,000

$$Z_{\text{Pump}} = 2100 \left(\frac{W_{\text{Pump}}}{10} \right)^{0.26} \left(\frac{1 - \eta_{\text{Pump}}}{\eta_{\text{Pump}}} \right)^{0.5} + 500 \left(\frac{W_{\text{Pump}}}{10} \right)^{0.87} \left(\frac{1 - \eta_m}{\eta_m} \right) \quad (11)$$

$$Z_{\text{compressor}} = 7364 \dot{m}_{12} PR_{\text{com}} \left(\frac{\eta_{\text{com}}}{1 - \eta_{\text{com}}} \right)^{0.7} \quad (12)$$

$$Z_{\text{Turbine}} = \frac{479.34 * \dot{m}_{\text{ORC}}}{0.92 - \eta_{\text{tur}}} \ln(PR_{\text{tur}}) * (1 + \exp(0.036T_{\text{RC4}} - 54.4)) \quad (13)$$

$$Z_{\text{Cooling Tower}} = 3950.9 (\dot{m}_{\text{CW}}^{0.58729} \times 10^x) \quad (14)$$

where

$$x = 0.0032091A.B - 0.026719A + 0.043654B - 0.1026 \quad (15)$$

A is the difference of the cooled water temperature and the wet bulb temperature, B is the temperature difference between the inlet and outlet water of the cooling tower.

The capital cost for the generator, absorber, rectifier, evaporator, condenser, RHX, intercooler, boiler and regenerator as heat exchangers is calculated by (Seyfour et al. 2018):

$$Z_k = Z_{k,R} \left(\frac{A_k}{100} \right)^{0.6} \quad (16)$$

$Z_{k,R}$ depicts the reference component cost as presented Table 1.

The heat transfer area of heat exchangers is:

$$A_k = \frac{Q_k}{U_k \text{LMTD}_k} \quad (17)$$

LMTD and U are the logarithmic mean temperature difference and the overall heat transfer coefficient of each heat exchanger, respectively. The overall heat transfer coefficient

of each heat exchanger is calculated by the authors as mentioned in Ref (Seyfour et al. 2018).

All investment costs become up to date to a common year by the cost index that its value provided in the reference (Cepci 2018). So the investment cost of each component is calculated by the Eqs. (9–17), and then, the total investment cost of the system is estimated. After that the total product cost is assessed (by Eq. 6) and is used for comprising in various systems.

3 Improved HGAX Cycles Description

When the temperature of the evaporator is low, the pressure of the evaporator is low too; the compressor should provide high-pressure ratio since the absorber pressure stays high, because as mentioned before in Sect. 1, if the absorber pressure becomes very low the GAX heat exchanger could not be used. The high pressure ratio of the compressor makes high-temperature fluid after compress process. The heat of the working fluid exiting of the compressor can be recovered in the system; furthermore, by utilizing more than one compressor the high temperature of the compressed fluid can be reduced and therefore the HGAX cycle can be improved. In this study by utilizing these improvements of HGAX cycle, four new configurations of the refrigeration system are suggested to produce cooling at low temperatures. The energy efficiencies, the exergy efficiencies and the circulating water of the cooling towers of these configurations are compared with the base system. The schematic diagrams of these configurations are presented in Fig. 2. The Rankine cycle in all configurations is the same, but the HGAX cycle is varied from one configuration to another.

3.1 Performance of Configuration 1

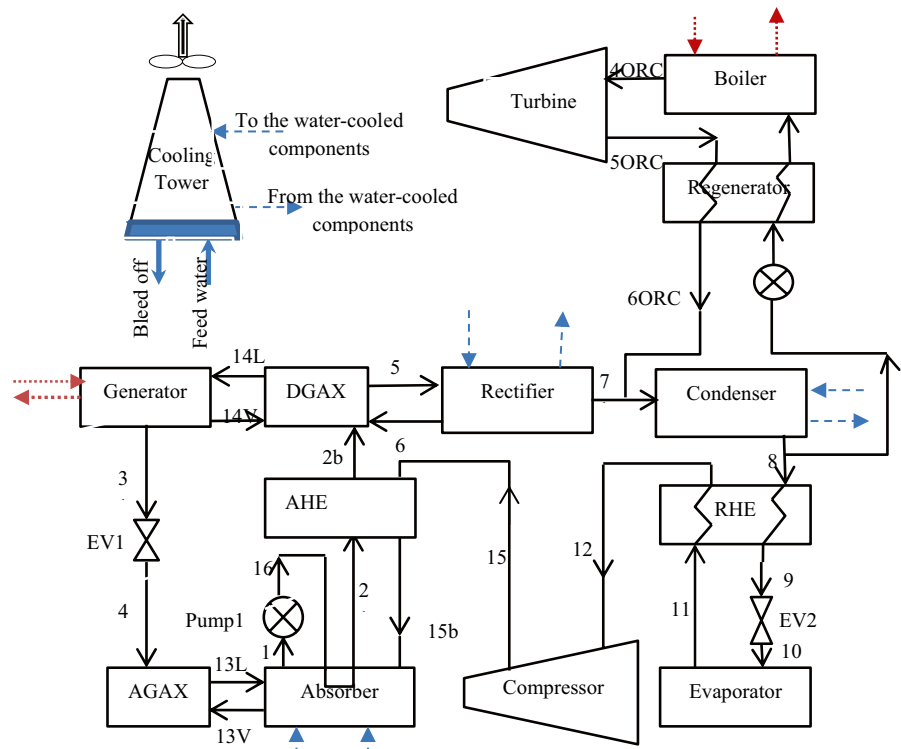
To utilize the heat of the working fluid exiting the compressor, in this configuration an additional heat exchanger is added to the system as preheater of the solution entering the desorber GAX; this weak solution is heated by the mixture coming from the compressor as shown in Fig. 2a.

$$\dot{m}_{15}(h_{15} - h_{15b}) = \dot{m}_2(h_{2b} - h_2) \quad (18)$$

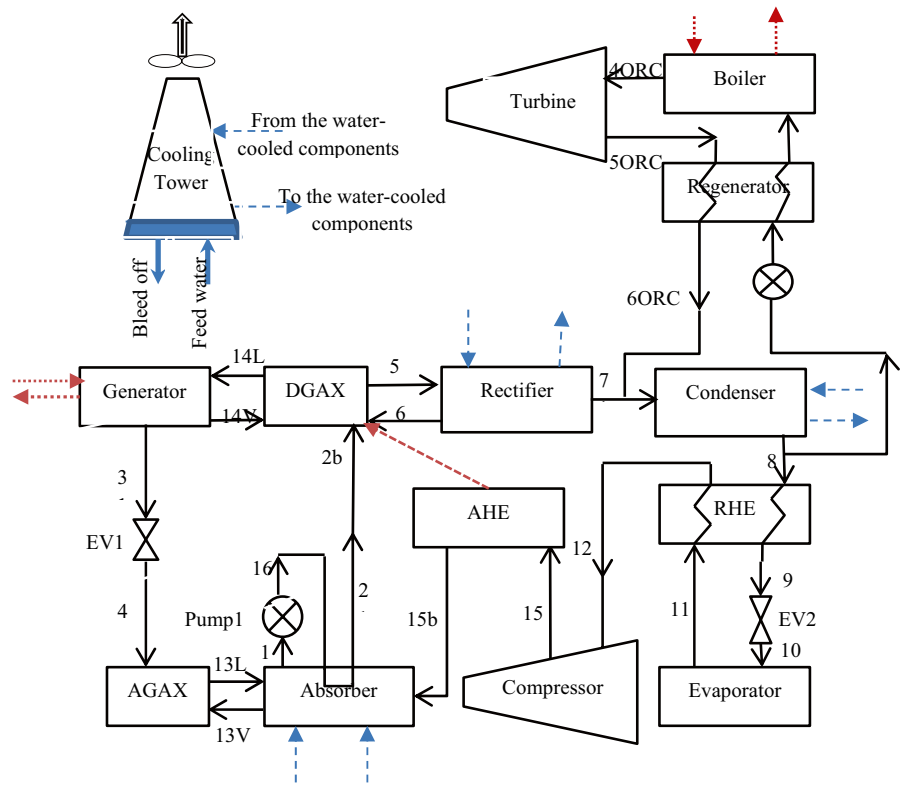
3.2 Performance of Configuration 2

When the heat required for the desorber GAX (DGAX) is higher than the available heat at the absorber GAX (AGAX), the second improvement of the HGAX cycle in this study can be applied. In the second improved cycle, the additional heat is given to DGAX by the working fluid exiting the compressor in an auxiliary heat exchanger (Fig. 2b).

Fig. 2 Schematic diagrams of four improved configurations for proposed refrigeration system

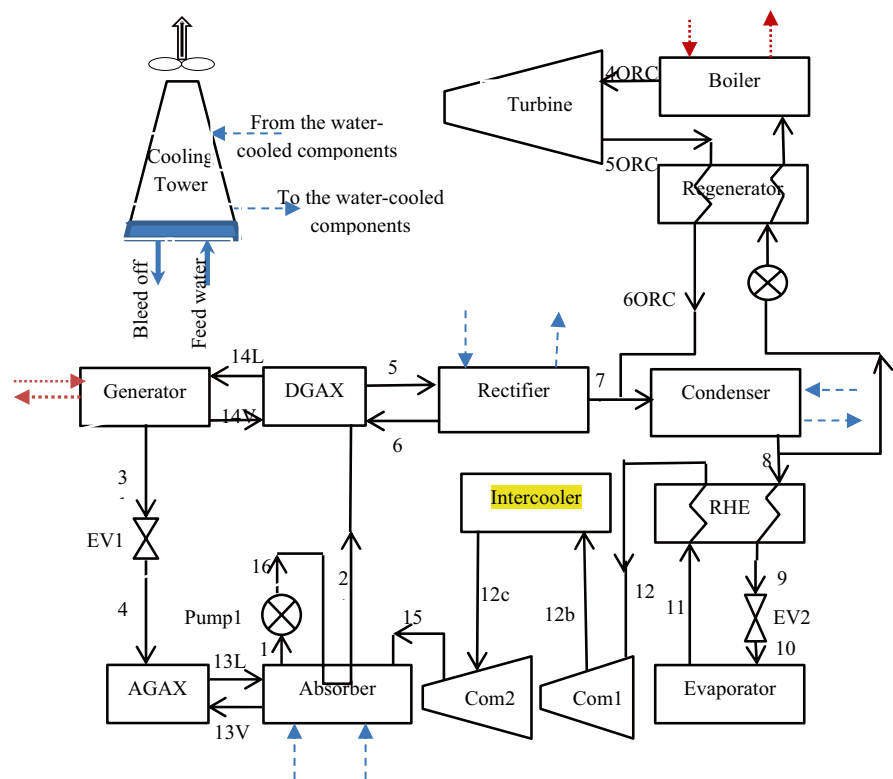


a) Schematic diagram of Configuration 1: RC and HGAX with preheater

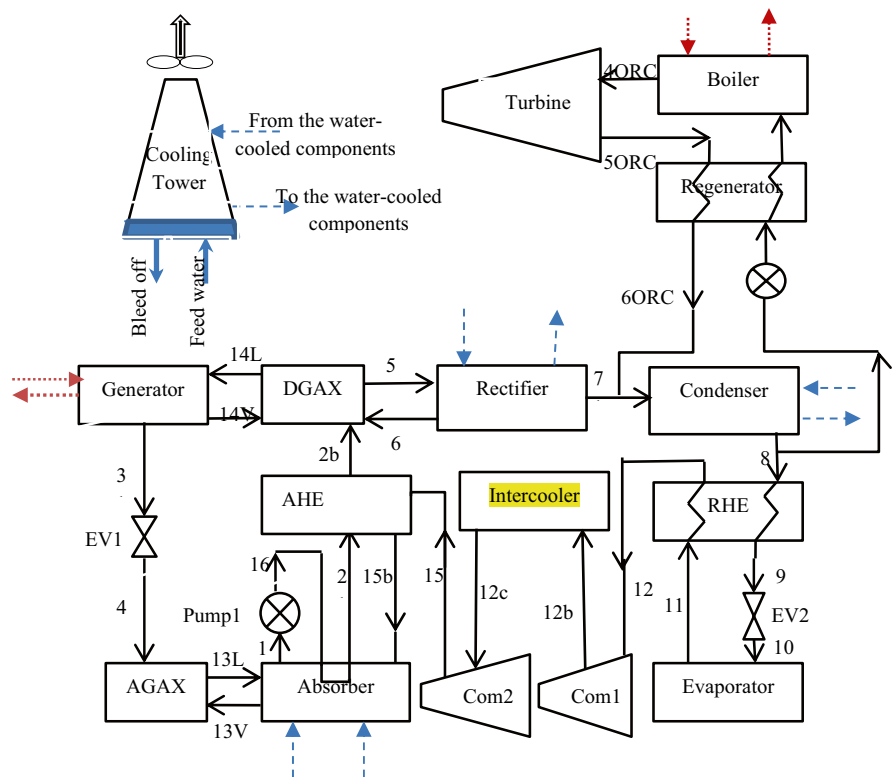


b) Schematic diagram of Configuration 2: RC and HGAX with auxiliary heat exchanger

Fig. 2 (continued)



c) Schematic diagram of Configuration 3: RC and HGAX with two compressors



d) Schematic diagram of Configuration 4: RC and HGAX with auxiliary heat exchanger and two compressors

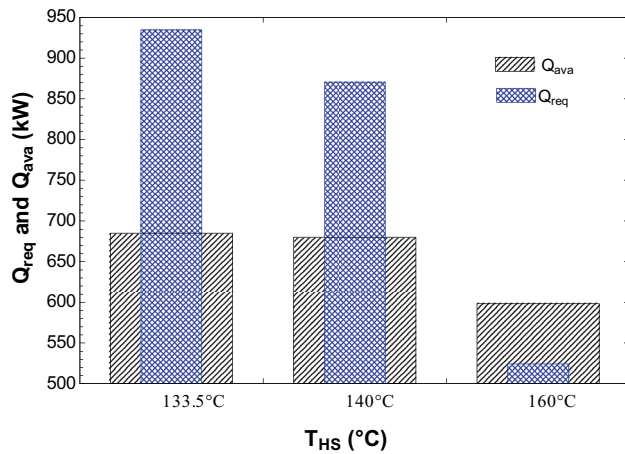


Fig. 3 Heat required for the DGAX and heat available in the AGAX of the base system for various heat source temperatures

$$\dot{Q}_{AHX} = \dot{Q}_{\text{requirement heat of the DGAX}} - \dot{Q}_{\text{available heat of the AGAX}} \quad (19)$$

$$\dot{Q}_{AHX} = \dot{m}_{15}(h_{15} - h_{15b}) \quad (20)$$

Figure 3 shows that in the basic HGAX cycle the heat requirement is higher than the heat available for heat source temperatures of 133.5 and 140 °C, but at higher heat source temperatures the available heat is sufficient to provide the GAX required heat; therefore, Configuration 2 is not any better for high-temperature heat sources.

3.3 Performance of Configuration 3

Another improvement of the HGAX cycle in this study is an additional compressor to reduce the fluid temperature after compression process and also to reduce the power requirement of the HGAX cycle. As shown in Fig. 2c, the working fluid is cooled between two compressors by cooling water at the intercooler.

3.4 Performance of Configuration 4

This configuration is ensemble of Configuration 1 and Configuration 3, the HGAX cycle in this configuration has two compressors and also a preheater of the DGAX (Fig. 2d).

These configurations are studied for the first time in this research; therefore, to validate them the HGAX cycle and Rankine cycle should be validated separately. These cycles were validated in the previously authors work in (Seyfour et al. 2018), so the validation results are not showed here another time.

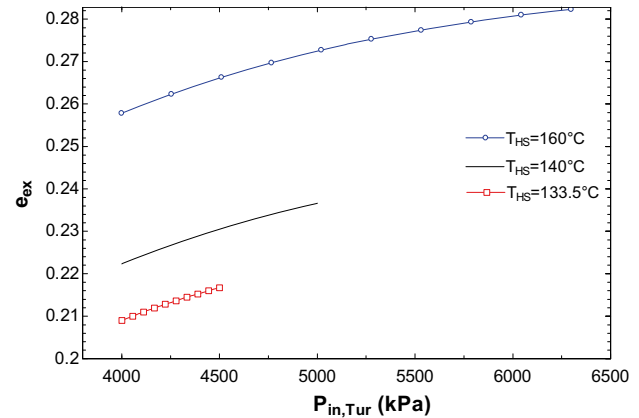


Fig. 4 The effect of inlet turbine pressure on the exergy efficiency for various heat Source temperatures. ($P_a = 310, 270, 165$ kPa for heat Source of 133.5, 140 and 160 °C, respectively)

4 Results and Discussion

In this study, four configurations of a refrigeration system are compared, while they produce 1000 kW cooling at -55 °C.¹ The hot water enters the generator and boiler at 133.5 °C² and/or 140 °C and/or 160 °C. The turbine inlet temperature and the generator temperature are considered, respectively, for each heat source temperature as: 125 °C, 132 °C and 152 °C with a proper minimum temperature approach. The temperature of the cooled water coming from the simulated cooling tower was calculated as 24.41 °C. The turbine inlet pressure, absorber pressure and the mean pressure between two compressors are chosen to have maximum exergy efficiency, because the second law of thermodynamics provides more meaningful appraisal than the first law.

Figure 4 shows the variation of exergy efficiency against turbine inlet pressure for the base system. It can be seen that at each heat source temperature the exergy efficiency is increased with an increase in turbine inlet pressure. The upper bound for the turbine inlet pressure is limited by temperature approach at the regenerator exit. The turbine inlet pressure at which the exergy efficiencies are maximized is 4500, 5000 and 6300 kPa, respectively, for heat source temperatures of 133.5, 140 and 160 °C.

According to Fig. 5, higher exergy efficiency is associated with the minimum absorber pressure in all heat source temperatures. The minimum absorber pressure as much as possible is dependent on the temperature of the absorber cooling water and also dependent on the evaporator and generator pressures. Therefore to have higher exergy efficiency the absorber pressures are chosen as 310, 270 and 165 kPa,

¹ Same as cooling temperature in references [3, 20].

² Same as hot water input in reference [12].

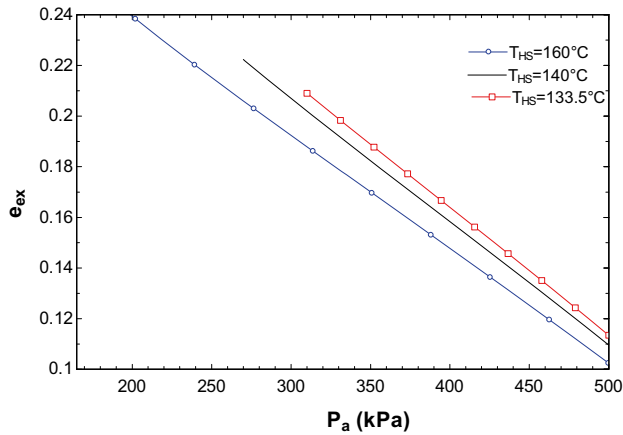


Fig. 5 The effect of the absorber pressure on the exergy efficiency for various heat source temperatures. ($P_{in,tur} = 4000$ kPa)

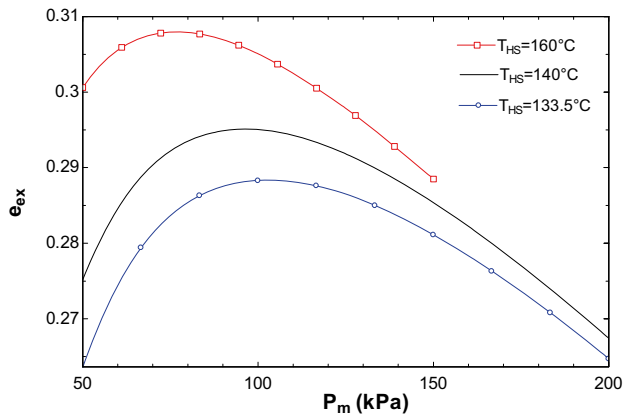


Fig. 6 The effect of mean pressure between two compressors on the exergy efficiency for various heat source temperatures of Configuration 3

respectively, for the heat source temperatures of 133.5, 140 and 160 °C.

To choose the optimum mean pressure between two compressors (in the Configurations 3 and 4), the effect of mean pressure on the exergy efficiency is illustrated in Fig. 6 for various heat source temperatures. It can be seen that for each heat source temperature, the maximum exergy efficiency occurs in a unique mean pressure. The pressure ratios in the compressors have significant effect on the performance of the system. To find the accurate value of optimum pressure, the EES software direct search method is used and the optimum pressures are chosen as 102.7, 96.38, 76.75 kPa for the heat source temperatures of 133.5, 140 and 160 °C, respectively.

Table 2 gives the calculated turbine inlet pressure, absorber pressure and mean pressure between two compressors corresponding to the maximum exergy efficiency.

Table 2 Optimized variables of the system using for comparison of four configurations

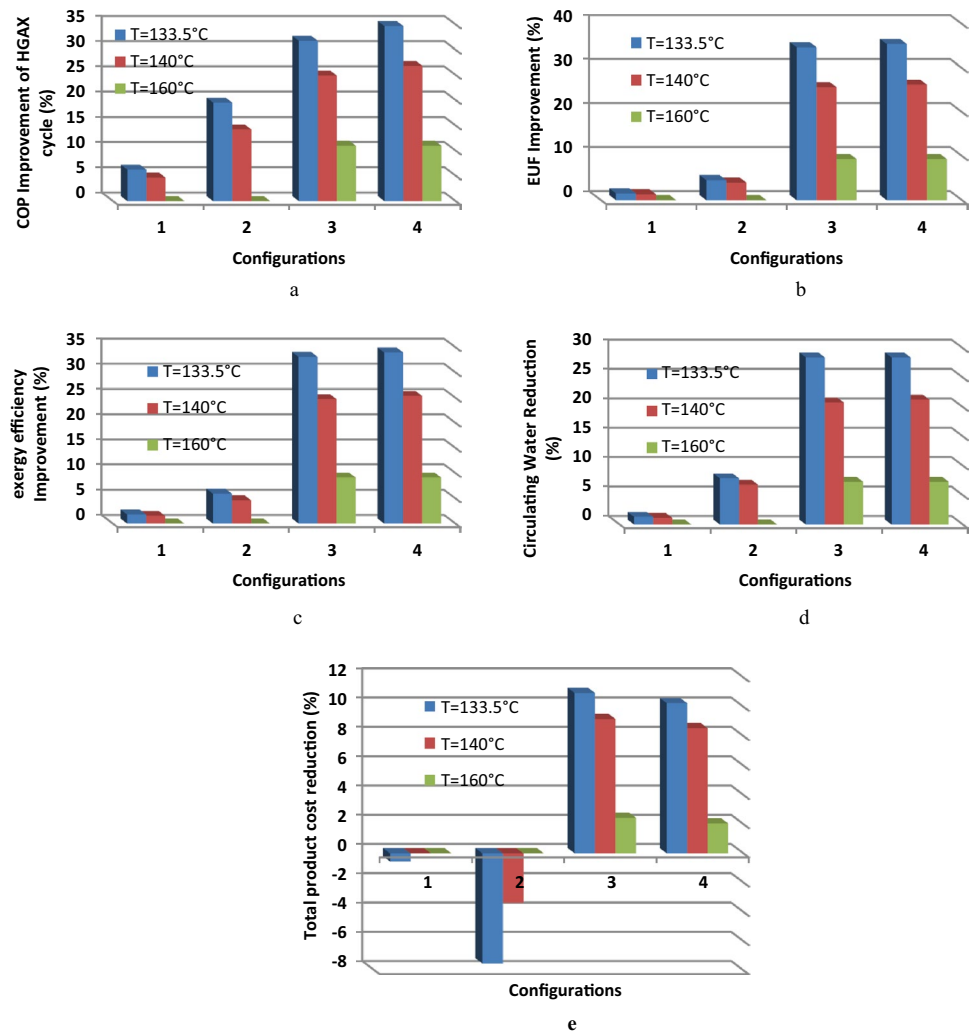
Decision variable	$T_{HS} = 133.5$ °C	$T_{HS} = 140$ °C	$T_{HS} = 160$ °C
T_G (°C)	125	132	152
$T_{in,tur}$ (°C)	125	132	152
$P_{in,tur}$ (kPa)	4500	5000	6300
P_a (kPa)	310	270	165
P_m (kPa) (for Configurations 3 and 4)	102.7	96.38	76.75

Table 3 Thermodynamic parameters for four configurations at various heat source temperatures

	$T_{HS} = 133.5$ °C	$T_{HS} = 140$ °C	$T_{HS} = 160$ °C
Configuration 1			
COP of the HGAX cycle	0.6933	0.6857	0.5876
EUF	0.1644	0.1903	0.2564
Exergy efficiency	0.2206	0.2402	0.2823
Circulating water (kg/s)	168.5	148.8	116.6
Total product cost (\$/hr)	14.43	12.77	9.793
Configuration 2			
COP of the HGAX cycle	0.7801	0.7487	–
EUF	0.1693	0.1953	–
Exergy efficiency	0.2296	0.2476	–
Circulating water (kg/s)	158.3	141	–
Total product cost (\$/hr)	15.43	13.2	–
Configuration 3			
COP of the HGAX cycle	0.8602	0.8186	0.6521
EUF	0.2181	0.236	0.2802
Exergy efficiency	0.2884	0.2951	0.308
Circulating water (kg/s)	132.9	124.6	108.7
Total product cost (\$/hr)	12.78	11.6	9.745
Configuration 4			
COP of the HGAX cycle	0.8793	0.8313	0.6521
EUF	0.2193	0.2371	0.2802
Exergy efficiency	0.2903	0.2966	0.308
Circulating water (kg/s)	132.2	124.1	108.7
Total product cost (\$/hr)	12.88	11.68	9.783

The COP of the HGAX cycle, EUF, exergy efficiency and total product cost of the overall system and the

Fig. 7 COP improvement of the HGAX cycle, EUF, and η_{ex} improvement, water consumption and total product cost reduction with respect to the corresponding parameters of the base system for four configurations at three heat source temperatures



circulating water of the cooling tower for three different heat source temperatures are presented in Table 2 for each configuration.

According to Table 3, the performance parameters of Configurations 3 and 4 are the same for the heat source temperature of 160 °C; the EUF and the exergy efficiency of Configurations 3 and 4 are about 0.28 and 0.31, respectively, and the performance parameters of Configurations 3 and 4 are better than two other improved configurations at all heat source temperatures. It is also evident from Table 3 that by increasing the heat source temperature, the COP of the HGAX cycle, the total product cost and circulating water of the cooling tower decrease, but EUF and exergy efficiency increase in all configurations.

The improvements of the COP of the HGAX cycle, EUF and exergy efficiency of four configurations in comparison with the base system are displayed in Fig. 7a–c. The reduction of circulating water in the cooling tower and total product cost in comparison with that of the base system are presented in Fig. 7d, e.

Figure 7 shows that the use of a preheater to preheat the input fluid of the DGAX improves the COP of HGAX cycle by about 6% for heat source temperature of 133.5 °C, but this preheater does not have significant effect on the energy and exergy efficiencies of the overall system. According to Fig. 7a–c the COP of the HGAX cycle, EUF and exergy efficiency improvements of Configurations 3 and 4 are higher than Configurations 1 and 2. It means that utilizing two compressors has more influence on the performance of the system than using a preheater or an auxiliary heat exchanger. From Fig. 7e, it is obtained that at Configuration 1, total product cost is the same as the base system and Configuration 2 increases the expenditures of the system, but the Configurations 3 and 4 reduce the product cost of the system.

The low performance differences between Configurations 3 and 4 show that using the preheater at the HGAX cycle with two compressors is not beneficial. It is also evident that with increasing the heat source temperature, the percentages of improvement of all configurations in comparison with the base system decrease; however, as mentioned before, for

the heat source temperature of 160 °C, the use of preheater and auxiliary heat exchanger does not have any effect on the HGAX cycle.

For the heat source temperature of 133.5 °C, the COP of the HGAX cycle in Configuration 3 is about 34% higher than that of the base system, the EUF and exergy efficiency of this system are, respectively, 35% and 33% higher than the base system. The circulating water in this configuration is 26% lower than the base system, and total product cost reduces 11% over the base system.

5 Summery and Conclusions

A comprehensive study is conducted to compare four new configurations of low-temperature refrigeration system. The system is consisted of two coupled cycles: a HGAX refrigeration cycle and a Rankine power cycle. Ammonia–water mixture is used as the working fluid of the system, and a wet cooling tower is utilized to provide the cooling water requirement. The turbine inlet temperature and pressure, absorber pressure and mean pressure between two compressors (Configurations 3 and 4) were calculated to yield the maximum exergy efficiency. The COP of the HGAX cycle, EUF, exergy efficiency, total product cost and circulating water of the cooling tower corresponding to the maximum exergy efficiency are computed for all configurations. The results are presented for three heat source temperatures; the results for heat source temperature of 133.5 °C are as follows:

- Configuration 1, consisted of HGAX cycle with preheater, improved the COP of the HGAX cycle, EUF, and exergy efficiency by about 6.2%, 1.5% and 1.8%, respectively, and reduced cooling water requirement by 1.3%. Also, the total product cost of this configuration is almost equal to the base system.
- Configuration 2, consisted of HGAX with auxiliary heat exchanger, improved the COP of the HGAX cycle, EUF, and exergy efficiency by about 19.5%, 4.6% and 6%, respectively, and reduced cooling water requirement by 7.8%. Also, the total product cost of this configuration is about 7.5% higher than the base system.
- Configuration 3, consisted of HGAX with two compressors, improved the COP of the HGAX cycle, EUF, and exergy efficiency by about 31.8%, 34.7% and 33%, respectively, and reduced cooling water requirement by 28%. Also, the total product cost of this configuration is about 11% lower than the base system.
- Configuration 4, consisted of HGAX with preheater and two compressors, improved the COP of the HGAX cycle, EUF, and exergy efficiency by about 34.7%, 35.45% and 34%, respectively, and reduced cooling water require-

ment by 28%. Also, the total product cost of this configuration is about 10% lower than the base system.

It is concluded that using the preheater in HGAX cycle improves the performance of the HGAX cycle with low-temperature heat sources, but it is not so efficient for the whole system. It can be seen that Configuration 4 is the most efficient configuration, but the higher total product cost than the configuration 3 and the slight performance improvement means that using the preheater in the HGAX cycle with two compressors is not beneficial; therefore, it is reasonable that Configuration 3, the system with two compressors in HGAX cycle, to be considered as the optimum configuration from the view point of thermodynamic analysis, economic analysis and water consumption.

Also, a higher heat source temperature results in lower COP of the HGAX cycles and higher EUF and exergy efficiency for all configurations, but at higher heat source temperatures all configurations have less improvement with regard to the base system, so that at heat source temperature of 160 °C, the use of preheater and auxiliary heat exchanger does not have any effect on the system performance.

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